

IOP Conference Series: Materials Science and Engineering

PAPER • OPEN ACCESS

The numerical assessment of motion strategies for integrated linear motor during starting of a free-piston engine generator

To cite this article: M Razali Hanipah and Akhtar Razul Razali 2017 *IOP Conf. Ser.: Mater. Sci. Eng.* **257** 012054

View the [article online](#) for updates and enhancements.

Related content

- [Numerical assessment of residual formability in sheet metal products: towards design for sustainability](#)
Javad Falsafi, Emrah Demirci and Vadim.V. Silberschmidt
- [Free-piston reciprocating cryogenic expander utilizing phase controller](#)
Jeongmin Cha, Jiho Park, Kyungjoong Kim et al.
- [Performance analysis on free-piston Stirling cryocooler based on an idealized mathematical model](#)
Y X Guo, Y J Chao, Z H Gan et al.

The numerical assessment of motion strategies for integrated linear motor during starting of a free-piston engine generator

M Razali Hanipah^{1,3*} and Akhtar Razul Razali^{2,3}

¹Automotive Engineering Research Group (AERG), Faculty of Mechanical Engineering,

²Manufacturing Focus Group (MFG), Faculty of Mechanical Engineering,

³Automotive Engineering Centre (AEC),

Universiti Malaysia Pahang, 26600, Pekan, Pahang, Malaysia.

*Corresponding author email: mohdrazali@ump.edu.my

Abstract. Free-piston engine generator (FPEG) provides a novel method for electrical power generation in hybrid electric vehicle applications with scarcely reported prototype development and testing. This paper is looking into the motion control strategy for motoring the FPEG during starting. There are two motion profiles investigated namely, trapezoidal velocity and S-curve velocity. Both motion profiles were investigated numerically and the results have shown that the S-curve motion can only achieve 80% of the stroke when operated at the proposed motoring speed of 10Hz.

1. Introduction

A free-piston engine (FPE) operates on dynamic balance of forces, which produces linear reciprocation motion. In the crankshaft engine, the piston position is consistent and can be represented by a kinematic relationship between crank radius, connecting rod length and crank angle. On the contrary, an FPE has non-fixed piston stop positions (top-dead-centre (TDC) and bottom-dead-centre (BDC)) and its piston motion is not governed by any mechanical component. Further, due to the absence of the crank-slider mechanism, the fundamental principle of operation of this engine requires a new approach.

A popular configuration of the FPE is the free-piston engine generator (FPEG) which has attracted commercial interest in recent years (for review, see e.g. Hanipah, Mikalsen and Roskilly [1]). However, the key step towards commercialisation is overcoming the hardware and software challenges [2]. One of the key challenges is the piston motion which posed numerous issues such as start-ability, misfiring and unstable operation [3]. This paper outlines the piston motion characterisation approach using numerical investigations.

2. Prior research

The typical configuration of a free-piston engine is a single piston configuration. Primarily, for cyclic operation to be possible, a free-piston engine requires a bounce device to ensure the piston returns to initial top-dead-centre position for the next engine cycle.

Free-piston engine is said to be dynamically constrained as opposed to kinematically constrained crankshaft engine [4]. A dynamics constraint means the piston stop positions (TDC and BDC) is not consistent and its motion profile is not governed by any mechanical component as in crankshaft



engine. Further, due to the absence of the crank-slider mechanism, the fundamental principles of operation of this engine require new approach.

The most popular configuration of a free-piston engine generator (FPEG) is a dual-piston type configuration allows each cylinder to act as bounce chamber [5, 6] for cyclic operation to be possible. The dual-piston type is shown Figure 1, where two pistons are connected by a shaft integrated with permanent magnet assembly as a single moving mass called a translator. The translator oscillates as a result of alternate combustion event occurring in the opposing combustion chambers.

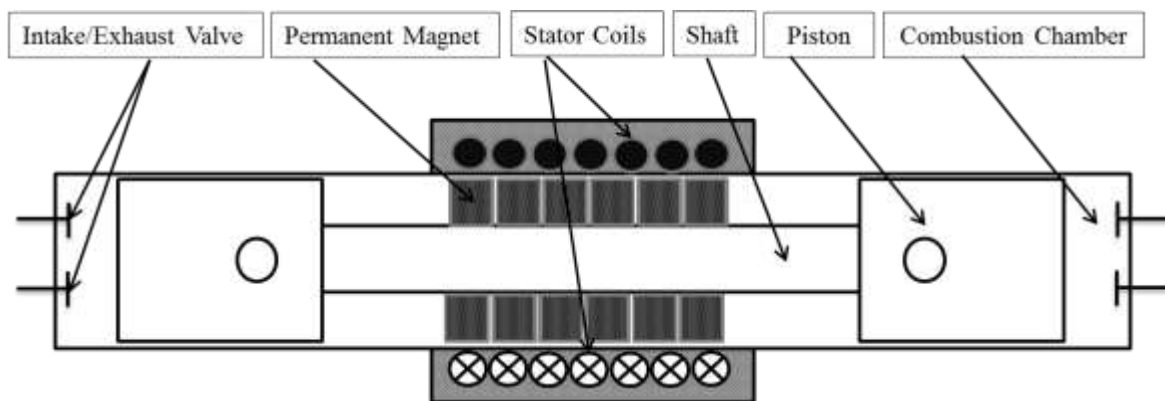


Figure 1. A dual-piston type free-piston engine generator (FPEG) configuration.

However, as oppose to conventional crankshaft engine, which can be cranked several times before it starts by using starter motor to drive the flywheel, the starting of an FPEG is arguably the key problem which can be addressed electrically [7-9]. Thus, any development of such engine must consider its starting scheme and strategy to ensure sufficient linear motor capability to overcome compression force at sufficient starting speed [6]. Further, the omission of flywheel creates a critical issue since the engine must obtain its energy from each stroke. Thus, researchers have suggested the integrated linear motor to provide assistance during misfire for continuous operation [6, 10].

In terms of engine cycle, FPEG must operate in two-stroke cycle although a complex and bulky four-stroke cycle version was theoretically shown to be possible [11]. The two-stroke version is simpler and thus, is widely adopted since the combustion occurs at every stroke to provide kinetic energy require for reciprocation thereby increasing its power output. Nevertheless, reported dual-piston configuration FPEG running on two-stroke seems to be plagued with scavenging issues [12, 13]. Some researchers eliminate this issue by using compressed air boost intake [5] and air-assisted direct injection [14].

The unique feature of free-piston engine generator (FPEG) lies in the piston dynamics. A single linear motion is converting the chemical energy from combustion process into kinetic energy of the moving mass and finally into electrical energy through the integrated generator. The electrical energy can then be stored in an energy storage system to be utilized for various applications such as for series hybrid electric vehicles and electrical generators.

Various researchers have applied Newton second law [10, 15-25] to model the dynamics of free-piston engine. It is limited to single piston and dual piston configuration since the opposed piston requires synchronizer mechanism, which introduce kinematic constraint to the pistons, which should be treated similar to crankshaft engine.

The principles of the numerical modelling of this paper is based on the dynamics model as established in [26] which is presented in the next section.

3. Piston motion numerical modelling

The piston motion profiles are the key aspect in controlling the energy input during starting of the free-piston engine generator (FPEG). This section establishes the dynamics model of the FPEG, which is based on the free-body diagram shown in Figure 2. The main forces comprise of:

- The in-cylinder pressure forces acting on both cylinders, F_{p_1} and F_{p_2}
- The frictional forces due to contact surfaces on the moving part of the engine, F_f
- The net force or inertial load of the moving mass, $m\ddot{x}$, (where \ddot{x} is the acceleration of the piston in the direction of motion and m moving mass).
- The electrical force acting on the permanent magnet assembly of the generator, F_{cog}
- The motoring force during starting, F_{mot}

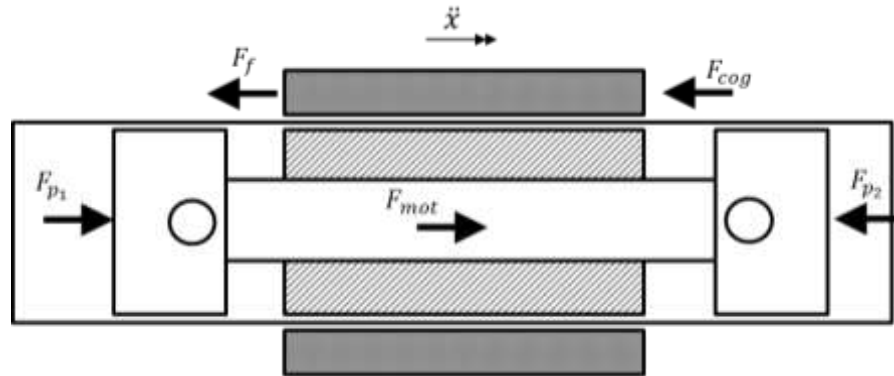


Figure 2. Free-body diagram of the dual piston free-piston engine generator dynamics model.

The dynamics equation can be obtained by applying Newton second law in the direction of acceleration as shown in Equation 1.

$$\sum F = F_{mot} + F_p - F_f - F_{cog} = m\ddot{x} \quad (1)$$

F_p is the resulting in-cylinder pressure forces acting on the translator can be expressed in the Equation 2:

$$F_p = F_{p_1} - F_{p_2} = (p_1 - p_2) \times \frac{\pi B^2}{4} \quad (2)$$

Where, p_1 and p_2 is the in-cylinder pressure in cylinder 1 and cylinder 2 respectively and B is the cylinder bore diameter.

Generally, there are three modes of free-piston engine generator operation as shown in Table 1. In motoring mode, which occurred during starting, the motoring force is the most dominant force. When combustion occurs, the combustion force $(F_p)_c$ becomes the most dominant. During this mode, motoring forces shall be gradually decreased until the sustainable reciprocation is achieved.

Table 1. Summary of the three main modes for a free-piston engine generator and the corresponding general equation of motion.

Mode	Equation of motion
Motoring	$\sum F = F_{mot} + (F_p)_m - F_f - F_{cog}$
Combustion	$\sum F = F_{mot} + (F_p)_c - F_f - F_{cog}$
Generating	$\sum F = (F_p)_c - F_f - F_{cog}$

The model developed in this paper is based on the design specifications presented in Table 2. The design has variable compression ratio capability of up to 20:1 but the maximum stroke is limited to 38mm for sufficient clearance between the cylinder head and the piston at TDC. The FPEG is expected to produce electrical power of 5kW at 50Hz by running in two-stroke cycle operation at a minimum of 2.8kW per-cylinder produced from combustion.

Table 2. The free-piston engine generator (FPEG) specifications.

Parameter	Value
Bore [mm]	50
Minimum/Maximum stroke [mm]	33/38
Minimum/Maximum Geometric compression ratio [-]	9.5:1 to 20:1
Moving mass [kg]	6.5
Minimum engine power/cylinder [kW/cyl] @50Hz	2.8
Maximum electrical power output [kW] @50Hz	5.0
Motoring speed [Hz]	10
Idling speed [Hz]	20
Generating speed [Hz]	50

There are two types motion profiles investigated in the numerical modelling:

- Trapezoidal velocity profile
- S-curve velocity profile.

The trapezoidal velocity profile is simpler and widely employed while the S-curve velocity profile, although complex, leads to smoother vibration-less motion. Figure 3 shows the modelled motion based on trapezoidal velocity profile. The translator undergoes constant acceleration at the beginning of the motion, zero acceleration when it reaches maximum velocity and steadily decelerated when it reaches the targeted stroke. The main problem with trapezoidal motion profile is the bumpy motion resulting from the infinite jerk characteristic to produce constant acceleration and deceleration values.

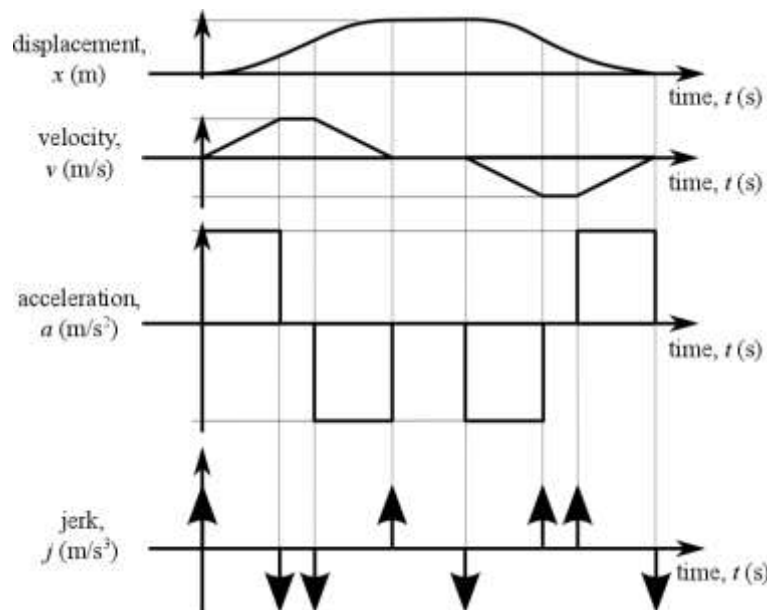


Figure 3. Trapezoidal velocity profiles with resulting acceleration and displacement profiles. The jerk is the derivative of acceleration.

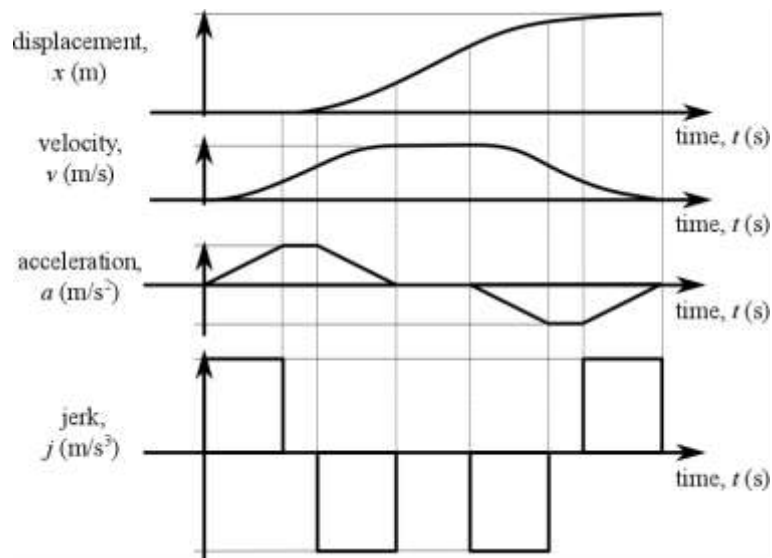


Figure 4. S-curve velocity profile with resulting acceleration and displacement profiles showing finite jerk characteristics.

Figure 4 shows the S-curve velocity profile with the resulting acceleration, and position profiles. In this profile, the jerk now has finite values due to the S-shaped velocity profiles. Both motion profiles are investigated in numerical model develop using Matlab and the results are discussed in the following section.

4. Results and discussion

The main aim of the paper is to investigate the piston motion resulting from two motion profiles commonly used for motion control presented earlier namely, trapezoidal and S-curve velocity profiles. Figure 5 shows a typical crankshaft piston velocity against piston position which produced higher piston velocity around TDC as compare to lower piston velocity for FPEG operating at similar engine cyclic speed.

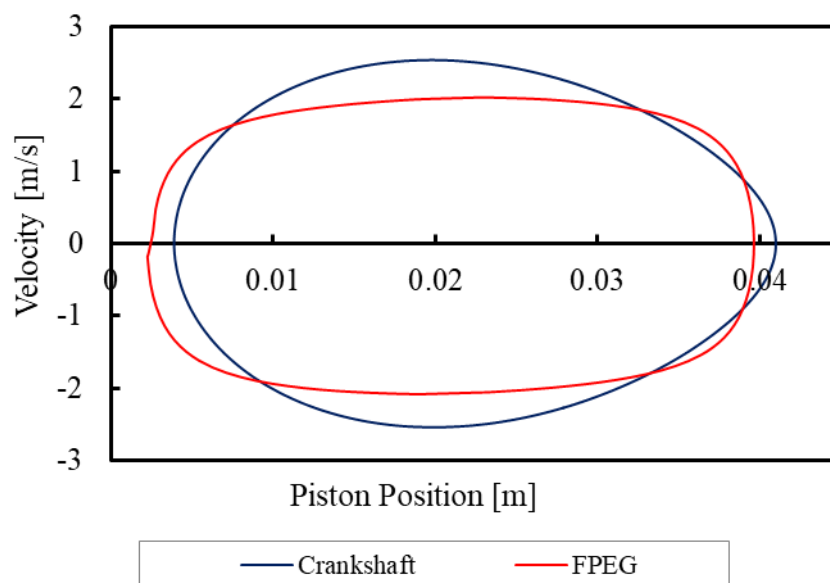


Figure 5. The typical velocity against piston position curves for crankshaft IC engine and free-piston engine generator (FPEG) used as baseline [26].

Figure 6 shows the piston velocity and acceleration at 10Hz motoring speed and 33mm stroke produced using trapezoidal velocity motion. There is no constant speed region since the piston is accelerated for 25ms and decelerated quickly so that the targeted stroke can be achieved. The typical trapezoidal velocity profile is observed with the maximum velocity occurs at mid-stroke, coincidental with the deceleration timing. This profile produced discontinuities for the jerk positions as presented in Figure 3 earlier.

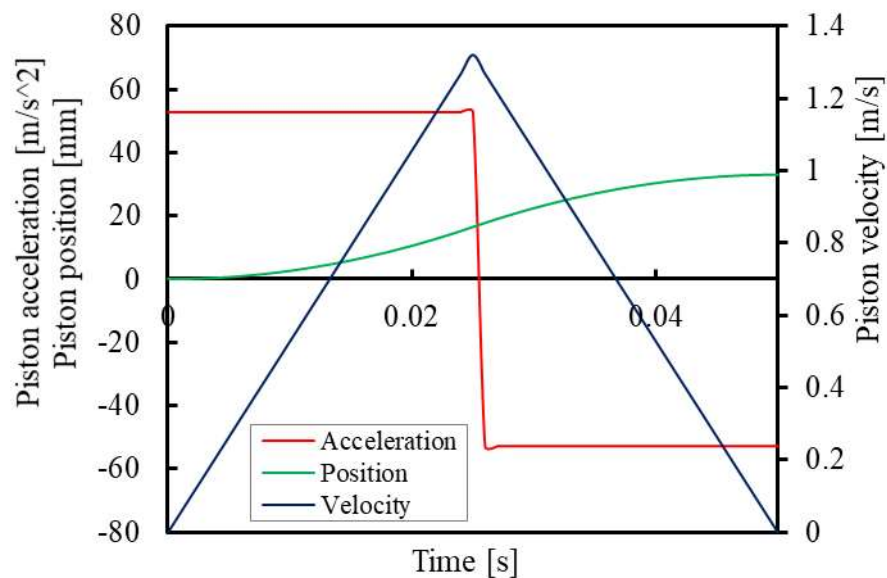


Figure 6. The acceleration and velocity produced at 10Hz motoring speed for the starting using trapezoidal velocity profile and its resulting piston position.

Figure 7 shows the resulting piston position using trapezoidal velocity profile at minimum stroke length of 33mm and maximum stroke of 38mm. Both targeted stroke positions are achieved within the cyclic duration at 10Hz. This is the main reason why this method is preferred widely for motion control applications.

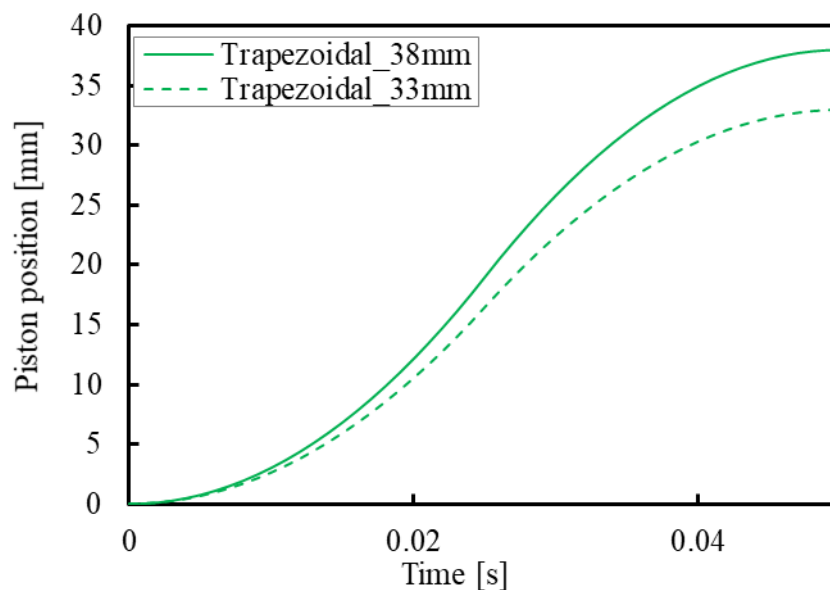


Figure 7. The piston position resulting from trapezoidal motion simulated at minimum and maximum stroke length.

Figure 8 shows the piston velocity and acceleration at 10Hz motoring speed and 33mm stroke produced using S-curve velocity motion. This motion produced favourable smooth acceleration transition for dual-piston type free-piston engine generator in order to reduce the cyclic vibration. However, the final position is 6mm shorter than the targeted stroke, which needs further tuning before this profile can be implemented.

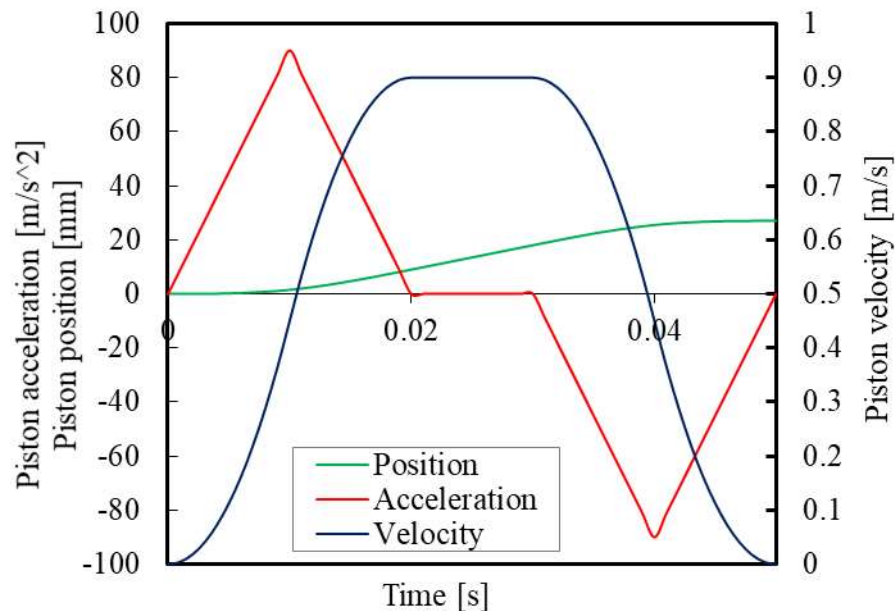


Figure 8. The acceleration and velocity produced at 10Hz motoring speed for the starting using S-curve velocity profile and its resulting piston position.

Figure 9 illustrates the severity of this situation where the stroke reduction is about 18% when S-curve motion profile is used. It can be observed that the piston motion has slower ascend during the first and last 10ms of the motion due to the S-shaped velocity profile which has contributed to the shorter stroke.

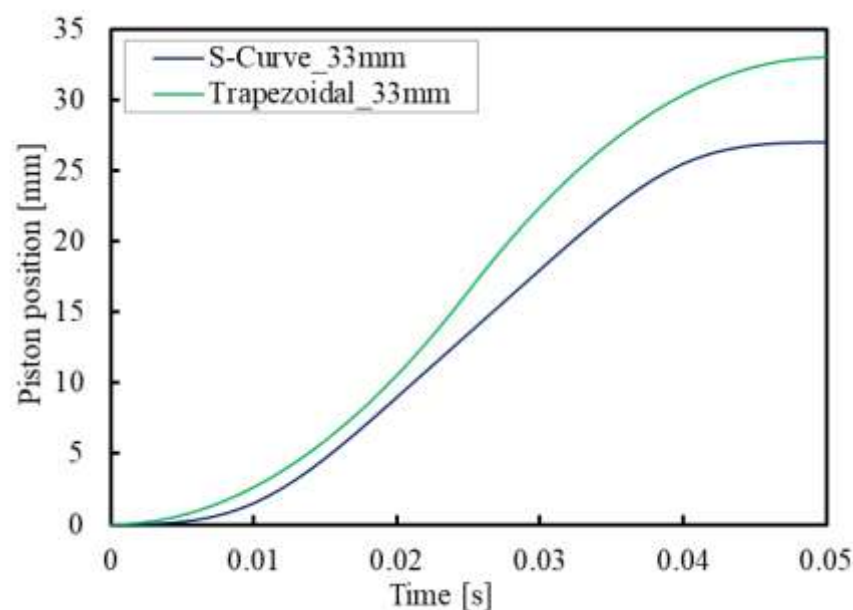


Figure 9. The piston position comparison between trapezoidal and S-curve motion simulated at 33mm.

5. Conclusions

In this paper, a numerical method is used to investigate two types of motion profiles, trapezoidal velocity and S-curve velocity for starting of free-piston engine generator. The trapezoidal motion profile is able to move the piston to the targeted stroke of 33mm and 38mm. However, the final position produced via S-curve motion profile is 18% shorter due to the slower ascend and descend during the first and final 20% of the cyclic duration. Further tuning of the model is necessary before the S-curve motion can be implemented in the free-piston engine generator starting strategy.

Acknowledgments

The author would like to thank Universiti Malaysia Pahang (www.ump.edu.my) for providing laboratory facilities and financial assistance under project no. RDU160384.

References

- [1] Hanipah M R, Mikalsen R and Roskilly A P 2015 *Appl. Therm. Eng.* **75** 493-503
- [2] Kock F, Heron A, Rinderknecht F and Friedrich H 2013 *MTZ Worldw* **74** 38-43
- [3] Jia B, Mikalsen R, Smallbone A, Zuo Z, Feng H and Roskilly A P 2016 *Applied Energy* **179** 1166-75
- [4] Aichlmayr H T 2002 *Design Considerations, Modeling and Analysis of Micro-Homogeneous Charge Compression Ignition Combustion Free-Piston Engines.*: UNIVERSITY OF MINNESOTA)
- [5] Nandkumar S 1998 *Two-Stroke Linear Engine*. In: *Department of Mechanical and Aerospace Engineering*, (Morgantown, West Virginia: West Virginia University)
- [6] Saiful Azrin M Z 2007 *Modeling, Simulation and Implementation of Rectangular Commutation for Starting of Free-Piston Linear Generator*. In: *Electrical and Electronic Engineering Department: Universiti Teknologi PETRONAS*)
- [7] Arof H, Eid A M and Nor. K M 2004 *Australasian Universities Power Engineering Conference (AUPEC 2004)*
- [8] Zulkifli S A, Karsiti M N and Aziz A R A 2008 Rectangular current commutation and open-loop control for starting of a free-piston linear engine-generator. In: *Power and Energy Conference, 2008. PECon 2008. IEEE 2nd International*, pp 1086-91
- [9] Zulkifli S A, Karsiti M N and A-Aziz A R 2009 Investigation of linear generator starting modes by mechanical resonance and rectangular current commutation. In: *Electric Machines and Drives Conference, 2009. IEMDC '09. IEEE International*, pp 425-33
- [10] Němeček P, Šindelka M and Vysoký O 2003 *IFAC Symposium on Advances in Automotive Control*.
- [11] Hunt B R, Lipsman R L and Rosenberg J M 2001 *A guide to MATLAB for Beginners and Experienced Users*: Cambridge University Press)
- [12] Goldsborough S S 2002 *Optimizing the Scavenging System for High Efficiency and Low Emissions: A Computational Approach*. In: *Department of Mechanical Engineering*, (Fort Collins, Colorado: Colorado State University)
- [13] Mao J L, Zuo Z X, Li W and Feng H H 2011 *Applied Energy* **88** 1140-52
- [14] Němeček P and Vysoký O 2006 *Proceedings of the 6th Asian Control Conference VOL.1*
- [15] Atkinson C M, Petreanu S, Clark N N, Atkinson R J, McDaniel T I, Nandkumar S and Famouri P 1999 *SAE International: Hybrid Vehicle Engines and Fuel Technology*. **1999-01-0921**
- [16] Goldsborough S S and Blarigan P V 1999 *SAE International* **1999-01-0619**
- [17] Houdyschell D 2000 *A Diesel Two-Stroke Linear Engine*. In: *Department of Mechanical and Aerospace Engineering: West Virginia University*) p 64
- [18] Johansen T A, Egeland O, Johannessen E A and Kvamsdal. R 2001 *Proc. American Control Conference*,
- [19] Fredriksson J and Denbratt I 2004 *SAE International* **2004-01-1871**
- [20] Johansen T A, Egeland O, Johannessen E A and Kvamsdal. R 2003

- [21] Shoukry E F 2003 *Numerical Simulation for Parametric Study of A Two-Stroke Compression Ignition Direct Injection Linear Engine*. In: *Department of Mechanical and Aerospace Engineering*, (Morgantown, West Virginia: West Virginia University)
- [22] Hansson J, Leksell M, Carlsson F and Sadarangani. C 2005
- [23] H.Xia, Y.Pang and M.Grimble 2006 *Hybrid Modelling and Control of a Free-Piston Energy Converter*. In: *International Conference on Control Applications*, (Munich, Germany)
- [24] Deutsch P and Vysoký O 2007 *MECCA Journal of Middle European Construction and Design of Cars* **VOL.5**
- [25] Mikalsen R and Roskilly A P 2008 *Appl. Therm. Eng.* **28** 589-600
- [26] Hanipah M R 2015 Development of a spark ignition free-piston engine generator. In: *School of Mechanical and Systems Engineering*, (Newcastle upon Tyne, United Kingdom: Newcastle University)